Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1988

The Computer Simulation of Oil-Flooded Single Screw Compressors

Wu Jianhua
Xi'an Jiaotong University

Jin Guangxi
Xi'an Jiaotong University

THE COMPUTER SIMULATION OF OIL-FLOODED SINGLE SCREW COMPRESSORS

Wu Jianhua and Jin Guangxi

Department of Power Machinery Engineering

Xi'an Jiaotong University, Xi'an, P.R.China

ABSTRACT

A computer model for prediction and analysis of an oil-flooded single screw compressor performance is developed. The model includes suction preheat, pressurization of the gas in the control volume during the suction closure process, flowage of gas and oil mixture through discharge port, leakage of gas and oil. It considers the effect of oil-injection rate, clearances and length of leakage paths on leakage, the effect of the discharge port area variation and the specific oil-injection rate on the discharge coefficient of the discharge port and the oil shear powers. Some calculated results are presented and compared with laboratory test.

SYMBOLS

A	area
C	coefficient, specific heat
G	mass flow rate unit area
h	specific enthalpy
k	ratio of C_p to C_v
L	length
m	mass, mass flow rate ($\dot{m}$)
p	pressure
R	gas constant, radius
T	temperature
U	velocity
V	volume
v	specific volume
W	velocity,width of leakage path, power ($\dot{w}$)
x	dryness
α	heat transfer coefficient
δ	clearance, thickness
μ	absolute viscosity
ρ	density
φ	rotation angle
ω	angle speed
τ	pressure ratio

Subscript

1	mainrotor
2	gaterotor
c	control volume
d	discharge
f	friction
g	gas
geo	geometry
in	flow in, the state of intake
l	liquid (oil)
oi	oil-injection
s	suction

INTRODUCTION

The single screw compressor is a rotary positive-displacement compressor used in industry for gas compression and refrigeration. More attention is paid to it because of the following inherent features : both the radial and axial pressure loads on the mainrotor are balanced; the gaterotors do no work; each groove in the mainrotor is used twice per revolution.

In order to analyse the internal operation process of compressors, improve the efficiency of the prototypes and predict the newly designed compressors performance, a computer simulation method of oil-flooded single screw compressors is developed. The model considers the complexity of geometry, the relations of the leakage flow velocities with clearances values and lengths of the leakage paths, sealing effect of oil-injected into the working space on the gas leakage, suction preheat, pressurization of the gas in the working space during the suction closure process, flowage of gas and oil mixture through discharge port in the discharge process and oil shear powers. The results are compared with test data and the efficacy of the computer model is shown.

PHYSICAL MODEL

Fundamental Assumptions and Equations

For analysis purpose the process closing the suction port is taken out and is defined as suction closure process in this paper. Clearances between parts form leakage paths. Nine leakage paths are defined in this paper, as shown in Figure 1. The fundamental assumptions used in the analysis are[1][2]

(1) Conditions in a groove will be the same as in the other grooves when they are at the appropriate phase angle.

(2) Pressure in suction and discharge chambers is constant. When backflow takes place, only the gas will flow back.

(3) Pressure, gas temperature and oil temperature are homogeneous throughout the working space at any instant. Potential and kinetic energies of the gas and oil in the working space are negligible.

(4) In suction closure, closed compression and discharge process, the control volume is adiabatic and the heat transfer between the gas and oil will be negligible.

(5) The oil and the gas never change phase. The gas behaves as an ideal gas.

(6) Only the oil leaks through the leakage paths except L1, L2 and L9.

From the point of view of thermodynamics, the leakage paths, inlet port and discharge port can be treated as the mass transfer paths from Path One to Path Eleven, however L1 and L2 are not within the control volume. Therefore, thermodynamic equations describing suction closure, closed compression and discharges process have the unitary form. Based on the assumptions mentioned above, the following equations are obtained

$$\frac{dm_{gc}}{d\varphi_2} = -\sum_{k=9}^{11} \dot{m}_{gk} / \omega_2 \quad , \quad \frac{dm_{1c}}{d\varphi_2} = -\sum_{k=3}^{11} \dot{m}_{1k} / \omega_2$$

$$\frac{dV_{gc}}{d\varphi_2} = \frac{dV_{geo}}{d\varphi_2} - \frac{1}{\rho_{1c}} \frac{dm_{1c}}{d\varphi_2}$$

$$\frac{dP_c}{d\varphi_2} = -K_g P_c \left(\frac{dV_{gc}}{V_{gc} d\varphi_2} + \sum_{k=9}^{11} \frac{\dot{m}_{gk}}{\omega_2 m_{gc}} \tau_{gk} \right)$$

$$\frac{dT_{gc}}{d\varphi_2} = -T_{gc} \left[(k_g - 1) \frac{dV_{gc}}{V_{gc}\, d\varphi_2} + \sum_{k=9}^{11} (K_g \tau_{gk} - 1) \frac{\dot{m}_{gk}}{\omega_2 m_{gc}} \right]$$

$$\frac{dT_{1c}}{d\varphi_2} = -T_1 \sum_{k=1}^{11} (\tau_{1k} - 1) \frac{\dot{m}_{1k}}{\omega_2 m_{1c}}$$

Suction Process

The changing of the gas states from intake port to the groove when which is just meshed by a gaterotor tooth will be investigated in this process. For normal oil-flooded single screw compressors, the pressure drop due to gas flowing is negligible and the pressure in the groove at the end of suction process will be equal to the pressure at intake port. The main factor that effects the performance of a compressor is suction preheat in the process. In this case, the intake gas receives heat from the leaked gas. The following equation describing the phenomena is obtained :

$$\dot{Q} + h_{gin}\,\dot{m}_{gd} + 2\sum_{k=7}^{9} h_{gk}\,\dot{m}_{gk} = h_{gs}\,(\dot{m}_{gd} + 2\sum_{k=7}^{9}\dot{m}_{gk})$$

where

$$\dot{Q} = \alpha_s\,\Delta T\,\dot{m}_{1r}/(\omega_1 R_1)\cdot(\dot{m}_{gd} + \dot{m}_{gr})$$

Suction Closure Process

This process models the entrance of the gaterotor tooth into the groove[1]. Since the moving speed of the gaterotor tooth in the groove or reducing rate of control volume has a certain value and the area of the inlet port is very small during the late process, the pressure in the groove at the end of the process is higher than the pressure in the suction chamber and the trapped gas mass is also greater than the mass calculated by P_{in}, T_{in}. The important point of the process in the model is the calculation of the gas mass rate which flows out of the groove. By applying the flow model through a nozzle, it can be given

$$\dot{m}_{g10} = C_s A_{in}\sqrt{\frac{2K_g}{K_g-1}\,\frac{p_c}{\rho_{gc}}\left[\left(\frac{p_{in}}{p_c}\right)^{\frac{2}{K_g}} - \left(\frac{p_{in}}{p_c}\right)^{\frac{K_g-1}{K_g}}\right]}$$

Discharge Process

The flow through the discharge port is simplified to a series of flows through variable area ratio orifices. Then, the following formulas can be obtained [3] [4]

$$G_o = C_d\sqrt{\frac{2\,|p_d - p_c|}{v_o}}$$

$$\dot{m}_{g11} = \begin{cases} A_d G_o x_d, & p_c \geq p_d \\ -A_d G_o, & p_c < p_d \end{cases}, \qquad \dot{m}_{11} = \begin{cases} A_d G_o (1-x_d), & p_c \geq p_d \\ 0, & p_c < p_d \end{cases}$$

Leakage and Oil Shear Power

In this paper leakage mass flow rates and frictional or viscous shear powers between relative moving parts are calculated by analysing flow in the clearence.

Leakages Through L1 and L2

It is indicated by the experiment that these leakage paths are main gas leakage paths.The inlet conditions of the leakage flow are the conditions in discharge chamber and the outlet pressure is equal to the suction pressure. According to the definitions these leakage paths are without the control volume. The leakages are assumed to be adiabatic oil-gas homogeneous flow, whose dryness is taken to be $C_x X_d$. Taking account of viscous frition the following equation can be derived [5]

$$1 - G_o x\frac{R_g T_g}{p^2}\left(1 - \frac{R_g/p}{C_o + G_o^2 x v_o R_g/p}\right) = -G_o^2\left[\frac{\lambda}{2}\,\frac{v_o}{2} - \frac{K_g-1}{K_g}\,\frac{\alpha(T_g - T_1)}{p\,\omega_o}\right]$$

The gas leakage mass flow rates through L8 and L9 are then

$$\dot{m}_{g1} = W_1\delta_1 G_{o1}\,x\,, \qquad \dot{m}_{g2} = W_2\delta_2\,(G_{o2} + u_2/2/v_o)\,x$$

Leakages Through L3 -L8

Based on fundamental assumpations, these leakage paths will be filled with oil. For adiabatic incompressible viscous flow between parallel flat plates shown in Figure 2, mass flow rate through the 2-2 cross section is

$$\dot{m} = \rho\left[\frac{U}{2} + \frac{\delta^2(p_2-p_1)}{12\,\mu L}\right] W\delta$$

frictional power

$$\dot{W}_f = \left(\frac{\mu U}{\delta} - \frac{1}{2}\frac{(p_2-p_1)\delta}{L}\right) LWU$$

temperature difference

$$\Delta T = \frac{p_2-p_1}{\rho c_1} + \frac{\dot{W}_f}{\dot{m} c_1}$$

Leakage Through L9

The sealing effect of the oil film is determined by the oil film thickness, oil viscosity, pressure difference, clearance value and relative velocity. In this model, it is assumed that the thickness $\delta = C_x X_d$. For the flow shown in Figure 3 the mass flow rate of oil and gas can be written respectively :

$$\dot{m}_{l7} = \rho_l W_7 \frac{\delta_l}{6(\delta_l + \delta_g \frac{\mu_l}{\mu_g})}\left\{3\,U_7\left(\delta_l + 2\delta_g\frac{\mu_l}{\mu_g}\right) + C_{ol}\left[(3\delta_7 + \delta_l)\delta_g\frac{\mu_l}{\mu_g} + \delta_l^2\right]\delta_l\right\}$$

$$\dot{m}_{g7} = \rho_g W_7 \frac{\delta_l}{6(\delta_g + \delta_l \frac{\mu_g}{\mu_l})}\left\{3\,U_7 + C_{og}\left[\delta_g^2 + 4\,\delta_l(4\delta_7 - \delta_l)\frac{\mu_g}{\mu_l}\right]\right\}$$

COMPUTER MODEL

The computer model presented by this paper is a FORTRAN 77 program that makes use of primary geometric variables, leakage path parameters, operation conditions, physical property of gas and oil, empirical coefficients and computer model parameters to simulate the operation process of oil-flooded single screw compressors and to calculate the external performance of compressors.

RESULTS

The operation conditions are : P_{in} =.1 0 MPa, T_{in} = 26.5°C, P_d =.8 0 MPa, Q_{oi} = 2.8×10^{-4} m^3/s , T_{oi} = 46.6°C , n = 2940 rpm.

The comparison of the results of the prototype performance test with the computer simulation was shown in Figure 4 to Figure 6. Since the effect of leakage on a compressor displacement is very small at lower discharge pressure, higher oil injection rate and smaller built-in volume ratio, the close agreement between the calculated and measured volumetric efficiency, as shown in Figure 4, indicates the calculation of suction preheat and suction closure is accurate.Figure 5 shows the comparison of the efficiencies at different discharge pressures and different clearances. The reason why the calculation values of the adiabatic efficiencies are lower than the measure values is that eddy dissipation and bearing frictional loss are not considered. Figure 6 shows the situation at different oil-injection rates.

Figure 7 shows the effect of oil-injection rate on discharge port flow resistance and leakage. Oil leakages rates through L3 to L9 are show in Figure 8. The result of the calculation shows that the contributions of suction preheat, leakages through L1 and L2 and discharge from grooves in suction closure process are larger. In this paper the theoretical displacement of compressors is calculated by the total groove volume rather than the pocket volume at the start of closed compression process.

The calculation of the oil shear powers indicates that the mesh clearances, the clearances between the mainrotor and the casing and the length of sealing cylinder on mainrotor are required to study further in order to reduce the oil shear powers of the prototypes.

CONCLUSION

A simulation model predicting oil-flooded single screw compressor performance has been presented and shown to give good agreement with test data. The model allows intake temperature, discharge pressure, oil-injection rate and clearance to be varied. It was shown that the effect of the simulation model for analysing oil-flooded single screw compressor performance was obvious.

REFERENCES

1. Bein, T. W. , and Hamilton, J.F., "Computer Modeling of an Oil Flooded Single Screw Air Compressor ", ICECP, 1982.
2. Boblitt, W.W., and Moore, J., " Computer Modeling of Single-Screw Oil-Flooded Refrigerant Compressor", ICECP , 1984.
3. Benedict, R. P. , "Fundamentals of Pipe Flow" John Wiley & Sons, Inc., New York, 1980.
4. Chisholm, D., "Prediction of Pressure Drop of Pipe Fittings during Two-Phase Flow", Inter. Congr. Refrig., 1975.
5. Wallis, G.B., " One-dimensional Two-phase Flow", Mc Graw-Hill, 1979.

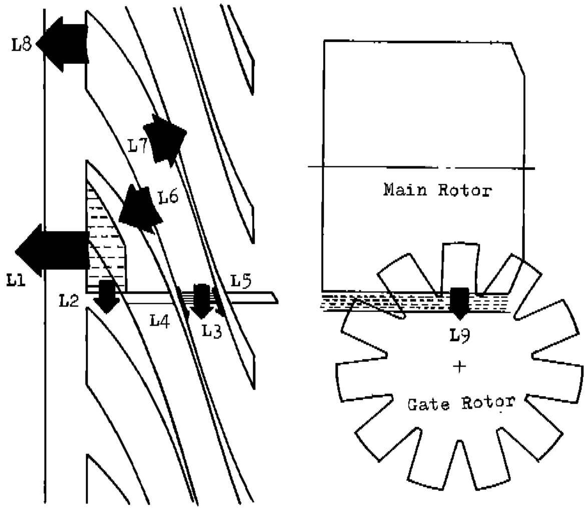

Fig. 1 Leakage Paths

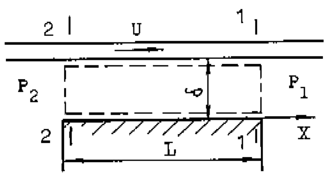

Fig. 2 Flow Model between Parallel Flat Plates

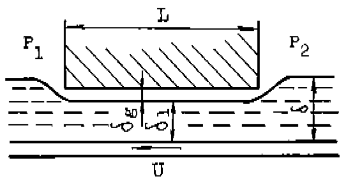

Fig. 3 Flow Medel Through L 9

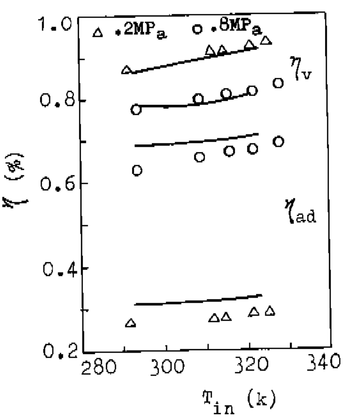

Fig. 4 Effect of Inlet temperature on Efficiency

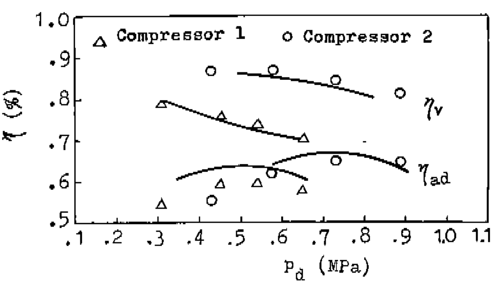

Fig. 5 Effect of Discharge Pressure on Efficiency

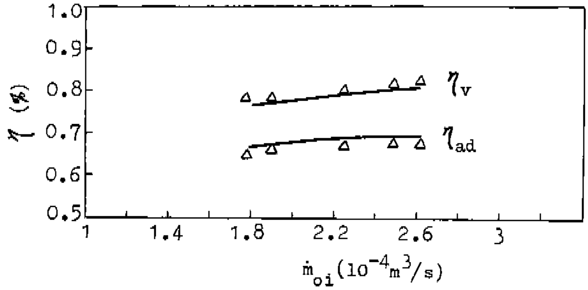

Fig. 6 Effect of Oil Mass Rate on Efficiency

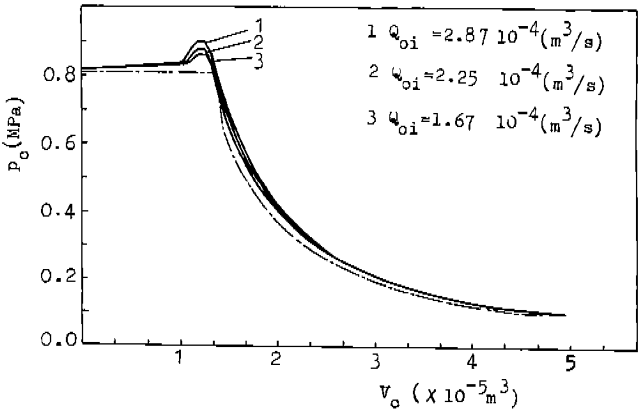

Fig. 7 Changing of Pressure in Control Volume

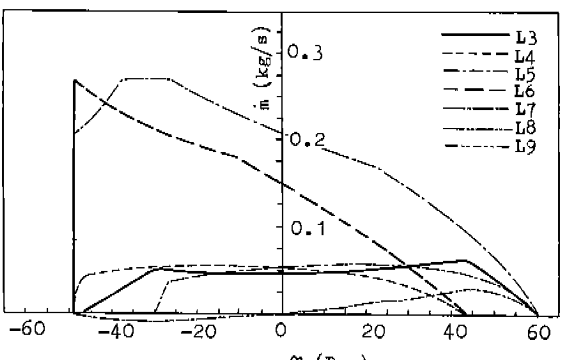

Fig. 8 Oil Mass Leakage Rate